5

Air Requirements

1 INTRODUCTION

The selection of a fan, blower or compressor is probably one of the most important decisions to be made in the design and specification of a pneumatic conveying system. It is often the largest single item of capital expenditure and the potential conveying capacity of the plant is dependent upon the correct choice being made. The output capability of the air mover is a major consideration in selection. The rating of the fan, blower or compressor is expressed in terms of the supply pressure required and the volumetric flow rate. Any error in this specification will result in a system that is either over-rated or is not capable of achieving the desired material flow rate [1].

For an existing pneumatic conveying system it is often necessary to check the performance, particularly if operating problems are encountered, or changes in material or conveying distance need to be considered. Here it is the conveying line inlet air velocity that is important. Since the determination of conveying line inlet air velocity and the specification of air requirements is so important for the successful operation of pneumatic conveying systems, all the appropriate models are derived and presented for reference purposes. In addition to the influence of pressure, temperature and pipeline bore, which are the primary variables, humidity is also considered.

1.1 Supply Pressure

The delivery pressure, or vacuum, required depends essentially upon the working pressure drop needed over the length of the conveying pipeline. The pressure drop across the gas-solids separation device can usually be neglected, but if a blow tank is used for feeding the material into the pipeline then an allowance for the pressure drop across the feeding device will have to be made. Consideration will also have to be given to the pressure drop in any air supply and extraction lines, and to the need for a margin on the value of conveying line pressure drop required to convey the material through the pipeline at the specified rate.

The magnitude of the conveying line pressure drop, whether for a positive or a negative pressure system, depends to a large extent on the conveying distance and on the solids loading ratio at which the material is to be conveyed. For short distance dilute phase conveying a fan or blower would be satisfactory, but for dense phase conveying or long distance dilute phase conveying, a reciprocating or screw compressor would be required. The pressure drop is also dependent upon the conveying gas velocity and a multitude of properties associated with the conveyed material.

1.2 Volumetric Flow Rate

The volumetric flow rate required from the fan, blower or compressor depends upon a combination of the velocity required to convey the material and the diameter of the pipeline to be used. Pipes and fittings are generally available in a range of standard sizes, but velocity is not so clearly defined.

For convenience the velocity at the end of the pipeline could be specified, for in the majority of cases compressors are rated in terms of 'free air delivered', and the pressure at the end of a pipeline, in positive pressure systems, in most applications, will be sufficiently close to atmospheric for this purpose. It is, however, the velocity at the start of the line that needs to be ascertained for design purposes. The problem is that air, and any other gas that is used for the conveying of materials, is compressible and so its density, and hence volumetric flow rate, is influenced by both pressure and temperature.

In negative pressure systems the air at the start of the conveying line is approximately at atmospheric pressure, and it decreases along the conveying line to the exhauster. For this type of conveying system, therefore, the minimum velocity that needs to be specified occurs at the free air conditions. Exhausters, however, are generally specified in terms of the volumetric flow rate of the air that is drawn into the air mover, and not free air conditions, and so it is essentially the same problem in evaluating air flow rates as with positive pressure conveying systems.

1.3 The Influence of Velocity

A conveying plant is usually designed to achieve a specified material flow rate. Material flow rate can be equated to the solids loading ratio and air mass flow rate.

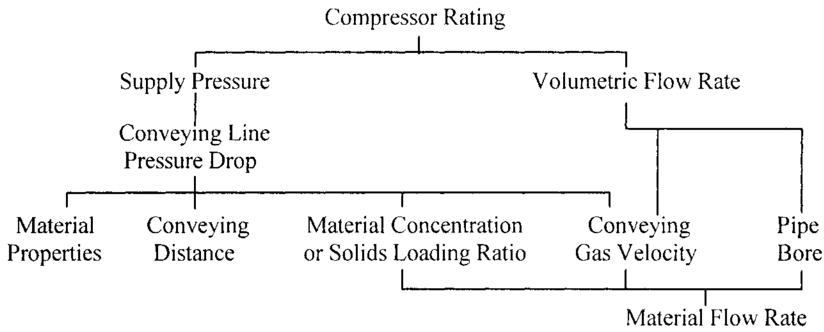

Figure 5.1 Parameters relating compressor rating with material flow rate.

The air mass flow rate is proportional to the volumetric air flow rate and this, in turn, is proportional to the air velocity and pipeline bore. Since these three parameters also have an influence on the compressor rating, it is extremely important that the correct air mover specification is made. The relationship between the various parameters that link the compressor rating and material flow rate is demonstrated with the path analysis shown in Figure 5.1.

Figure 5.1 also illustrates the importance of conveying air velocity in this relationship, as it influences both the supply pressure and the volumetric flow rate of the compressor. This helps to explain why conveying air is one of the most important variables in pneumatic conveying, and why it need to be controlled fairly precisely.

If, in a dilute phase conveying system, the velocity is too low it is possible that the material being conveyed will drop out of suspension and block the pipeline. If, on the other hand, the velocity is too high, bends in the pipeline will erode and fail if the material is abrasive, and the material will degrade if the particles are friable.

Velocity also has a major influence on the conveying line pressure drop, and hence on the mass flow rate of the material conveyed through a pipeline. The range of velocity, therefore, is relatively narrow, particularly in dilute phase systems, varying from a minimum of about 3000 ft/min to a maximum of around 6000 ft/min. This includes the compressibility effect, for the 3000 ft/min relates to the pipeline inlet and the 6000 ft/min relates to the pipeline outlet.

1.4 Air Movers

A wide range of air movers are available, but it is essential that the correct type of machine is chosen for the given duty. It is the characteristics of the air mover, in terms of the variation of the air flow rate with change of delivery pressure, at a given rotational speed, that are important for pneumatic conveying, as discussed in Chapter 3 on System Components.

In the majority of pneumatic conveying systems the air mover is driven at a constant speed, and design and operation is based on achieving a given conveying line inlet air velocity. If the material feed rate into the pipeline was to increase by 10%, there would be a similar increase in pressure demand. Even if the same air flow rate was delivered there would be a reduction in conveying line inlet air velocity because of the higher pressure. Motor sizes and air flow rate should be specified to take this type of fluctuation into account.

Ideally an air mover is required that will deliver the same air flow rate at the higher pressure. In practice a small reduction in air flow rate will result, and hence a further lowering in air velocity, and so this should be accommodated. With some air movers, a 10% increase in pressure demand will result in an even greater reduction in air flow rate. Air movers with this type of operating characteristic are unlikely to be acceptable.

The vast majority of the power required by a pneumatic conveying system is taken by the air mover. If a conveying system requires a large bore pipeline and a high pressure air supply, the power required is likely to be very high. Power requirements, and hence the cost of operation for pneumatic conveying, does tend to be much higher than for other conveying systems, particularly for materials conveyed in dilute phase. It must also be recognized that as a result of the high speed of compression, the air will be delivered at an increased temperature and so a decision will have to be taken on whether or not to cool the air.

1.5 Air Humidity and Moisture

Air is a mixture of gases. Oxygen and nitrogen are the main constituents, but it is also capable of absorbing a certain amount of water vapor. There is, however, a limit to the amount of water that air can hold in gaseous form as vapor. Relative humidity is a measure of the amount of moisture that air contains at a given pressure and temperature. It is expressed as a percentage. Relative humidity gives an indication of how dry the air is, and hence how much more vapor the air is capable of holding. 100% represents the limit for relative humidity and at this value the air is said to be saturated.

Specific humidity is a measure of how much moisture the air actually contains, and is usually expressed in terms of lb of water per lb of dry air. Relative humidity cannot rise above 100% and so if changes occur such that saturation conditions are exceeded, condensation will occur. The amount of moisture that air can support increases with increase in temperature and decreases with increase in pressure. Thus an increase in temperature will result in air becoming drier. A decrease in temperature will result in the relative humidity increasing.

If saturation conditions are reached then condensation will occur with any subsequent decrease in temperature. Compression of air is likely to result in condensation if there is no change in temperature. Across a compressor there is usually an increase in temperature, as well as pressure, and so at outlet the air is likely to be dry, as temperature generally has an over-riding effect in this situation.

1.6 Compressibility Effects

The volumetric flow rate of air required to convey a material through a pipeline can be evaluated from the cross sectional area of the pipeline and the air velocity required to convey the material. Consideration must be given, however, to the fact that air is compressible, and that it is compressible with respect to both pressure and temperature, and if the plant is not located at sea level, the influence of elevation may also have to be taken into account. As a result of the compressibility with respect to pressure, stepped bore pipelines are often employed and these are given due consideration.

Although it is air that is generally referred to, materials can be conveyed with any suitable gas. Constants are included in the equations that will correctly account for the type of gas used when evaluating the volumetric flow rate required. Air mass flow rate is also considered, as it is a useful working parameter, since its value remains constant in a pipeline, and is required for evaluating the solids loading ratio.

1.6.1 Conveying Air Velocity

For the pneumatic conveying of bulk particulate materials, one of the critical parameters is the minimum conveying air velocity necessary to convey a material. For dilute phase conveying this is typically about 3000 ft/min, but it does depend very much upon the size and size distribution, shape and density of the particles of the bulk material.

For dense phase conveying it can be as low as 600 ft/min, but this depends upon the solids loading ratio at which the material is conveyed and the nature of the conveyed material. If the velocity drops below the minimum value the pipeline is likely to block. It is important, therefore, that the volumetric flow rate of air, specified for any conveying system, is sufficient to maintain the required minimum value of velocity throughout the conveying system.

1.6.2 Material Influences

It should be noted that in evaluating conveying air velocities and volumetric air flow rates in pneumatic conveying applications, the presence of the material is disregarded in all cases, whether for dilute or dense phase conveying. The conveying air velocity is essentially the superficial value, derived simply by dividing the volumetric flow rate by the pipe section area, without taking account of any particles that may be conveyed.

In dilute phase conveying, and at low values of solids loading ratio, the influence of the conveyed material will have negligible effect in this respect. At a solids loading ratio of 100, however, the material will occupy approximately 10% of the volume at atmospheric pressure and so the actual air velocity will be about 10% higher. At increased air pressures and solids loading ratios the percentage difference will be correspondingly higher.

It would be a very complex and time consuming process to evaluate actual air velocities and so for convenience the superficial air velocity is universally employed. Critical values such as the minimum conveying air velocity and conveying line inlet air velocity are mostly derived from experience and experimental work. In such cases it is the superficial air velocity that is used.

As with the flow of air only in a pipeline, or single phase flow, the flow of a gas-solid mixture will also result if there is a pressure difference, provided that a minimum value of conveying air velocity is maintained. Material flow will be in the direction of decreasing pressure, whether it is a positive pressure or a vacuum conveying system. Since air is compressible, the volumetric flow rate of the air will gradually increase, from the material feed point at the start of the pipeline, to the material discharge point at the end of the pipeline. In a single bore pipeline the conveying air velocity will also gradually increase over the length of the pipeline.

This means that it is the value of the conveying air velocity at the material feed point, or the start of the pipeline, that is critical, since the value of the conveying air velocity will be the lowest at this point, in a single bore pipeline. In determining the necessary volumetric flow rate of air, therefore, it is the conditions prevailing at the start of the pipeline, in terms of pressure and temperature, that must be taken into account.

2 VOLUMETRIC FLOW RATE

Volumetric flow rate in ft^3/min has been chosen for use in all the mathematical models developed and on all graphical plots presented in this Handbook. Although it is not the basic fps unit, ft^3/min is more widely quoted in trade literature on blowers and compressors. It is also more compatible with the use of ft/min for air velocity. Inches have been used for all pipeline bore references.

2.1 Presentation of Equations

The majority of the equations that follow are presented in terms of both volumetric flow rate and conveying air velocity. The reason for this is the need to provide models that can be used for both the design of future systems and for the checking of existing systems. In the design of a system a specific value of conveying air velocity will generally be recommended, together with a pipe bore, and it is the value of volumetric flow rate that is required for specification of the blower or compressor.

In order to check an existing system it is usually necessary to determine the conveying air velocity for the particular conditions. In addition to providing the appropriate models for the evaluation of air requirements and conveying air velocities, graphical representation of these models is also presented. With programmable calculators and computers, models such as these can be handled quite easily and quickly.

Graphs, however, do have the advantage of showing visually the relative effects of the various parameters, and in some cases can be used very effectively to illustrate particular processes, and have been adopted widely in this Handbook. The main equations that are developed are additionally presented in SI units, and reference to the equivalent SI units for all symbols and dimensions is given in the Nomenclature at the end of this chapter.

2.2 The Influence of Pipe Bore

The diameter of a pipeline probably has the most significant effect of any single parameter on volumetric flow rate. The volumetric flow rate through a pipeline depends upon the mean velocity of flow at a given point in the pipeline and the pipe section area. The relationship is:

$$\dot{V} = \frac{C \times A}{144} \quad \text{ft}^3\text{/min} \qquad - - - - - - - - \quad (1)$$

where $\dot{V}$ = volumetric flow rate - $\text{ft}^3\text{/min}$
C = conveying air velocity - ft/min
and A = pipe section area - in^2

$$= \frac{\pi\, d^2}{4} \quad \text{for a circular pipe}$$

where d = pipe bore - in

so that

$$\dot{V} = \frac{\pi\, d^2 C}{576} \quad \text{ft}^3\text{/min} \qquad - - - - - - - - \quad (2)$$

or

$$C = \frac{576\, \dot{V}}{\pi\, d^2} \quad \text{ft/min} \qquad - - - - - - - - \quad (3)$$

A graphical representation of the above models is presented in Figure 5.2. This is a plot of volumetric air flow rate against conveying air velocity, with a series of lines representing the relationship for different sizes of pipe. Conveying air velocities from about 500 ft/min to 10,000 ft/min have been considered in order to cover the two extremes of minimum velocity in dense phase conveying and maximum velocity in dilute phase conveying, although velocities as high as 10,000 ft/min would not normally be recommended.

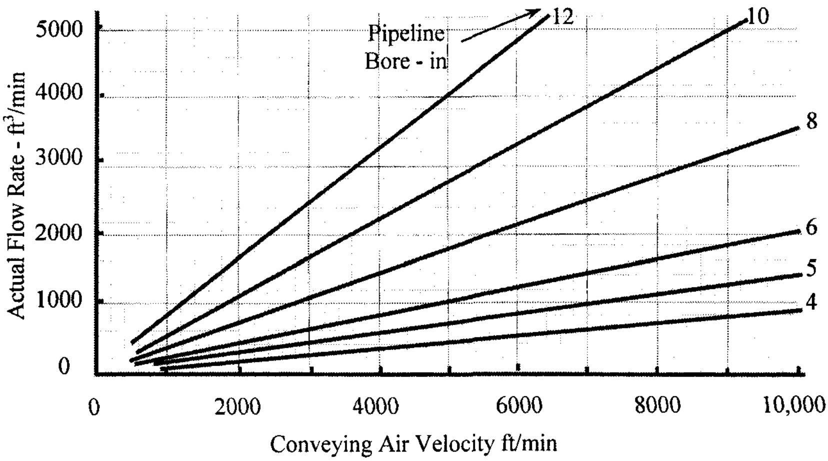

Figure 5.2 The influence of air velocity and pipeline bore on volumetric flow rate.

2.2.1 Reference Conditions

It should be noted that the volumetric flow rate on this graph is not related to any reference condition. It is the actual flow rate at any given condition of air pressure and temperature. Equations 5.1 to 5.3 and Figure 5.2 can be used either to determine the resulting velocity for a given flow rate in a given pipe size, or to determine the required volumetric flow rate knowing the velocity and pipe bore.

Blowers and compressors are usually rated in terms of 'free air delivered'. This means that the volumetric flow rate is related to ambient conditions for reference purposes - usually a pressure of 14·7 lbf/in^2 absolute and a temperature of 59°F (519 R). The influence of pressure and temperature on volumetric flow rate, and hence velocity, is discussed in the following sections.

2.2.2 Pipeline Influences

The air at the start of a conveying line will always be at a higher pressure than that at the end of the line because of the pressure drop necessary for air and material flow. Air density decreases with decrease in pressure and so, in a constant bore pipeline, the air velocity will gradually increase from the start to the end of the pipeline. The air mass flow rate will remain constant at any section along a pipeline, but as the rating of blowers and compressors is generally expressed in volumetric flow rate terms, then knowledge of the air mass flow rate is of little value in this situation.

2.3 The Ideal Gas Law

The relationship between mass and volumetric flow rate, pressure and temperature for a gas can be determined from the Ideal Gas Law:

$$144\ p\ \dot{V} = \dot{m}_a\ R\ T \qquad (4)$$

where p = absolute pressure of gas - lbf/in^2
$\dot{V}$ = actual volumetric flow rate of the gas at the pressure, p - ft^3/min
$\dot{m}_a$ = mass flow rate of gas - lb/min
R = characteristic gas constant - ft lbf/lb R
and T = absolute temperature of gas - R
= t °F + 460

Rearranging this gives:

$$\frac{p\ \dot{V}}{T} = \frac{\dot{m}_a\ R}{144}$$

For a given gas and constant mass flow rate:

$$\frac{p\ \dot{V}}{T} = constant$$

so that

$$\frac{p_1\ \dot{V}_1}{T_1} = \frac{p_2\ \dot{V}_2}{T_2} \qquad (5)$$

where subscripts $_1$ and $_2$ can relate to any two anywhere along the conveying pipeline

or in terms of 'free air conditions'

$$\frac{p_o\ \dot{V}_o}{T_o} = \frac{p_1\ \dot{V}_1}{T_1} \qquad (6)$$

where subscript $_o$ refers to reference conditions
usually p_o = 14·7 lbf/in^2 absolute
T_o = 519 R
and $\dot{V}_o$ = free air delivered in ft^3/min
and subscript $_1$ refers to actual conditions, anywhere along the conveying pipeline

2.3.1 Working Relationships

Substituting reference values into Equation 6 and rearranging gives:

$$\dot{V}_o = \frac{519 \times p_1}{14 \cdot 7 \times T_1} \times \dot{V}_1$$

$$= 35{\cdot}3 \times \frac{p_1 \dot{V}_1}{T_1} \quad \text{ft}^3\text{/min} \quad - - - - - - \quad (7)$$

$$= 2{\cdot}843 \times \frac{p_1 \dot{V}_1}{T_1} \quad \text{m}^3\text{/s} \quad - - - - - - \quad (7\ \text{SI})$$

or alternatively

$$\dot{V}_1 = 0{\cdot}0283 \times \frac{T_1 \dot{V}_o}{p_1} \quad \text{ft}^3\text{/min} \quad - - - - - - \quad (8)$$

$$= 0{\cdot}352 \times \frac{T_1 \dot{V}_o}{p_1} \quad \text{m}^3\text{/s} \quad - - - - - - \quad (8\ \text{SI})$$

2.3.2 Gas Constants

The constant, R, in Equation 4 has a specific value for every gas and is obtained from:

$$R = \frac{R_o}{M} \quad \text{ft lbf/lb R} \quad - - - - - - - - - - \quad (9)$$

where R_o = universal gas constant - ft lbf/lb-mol R
= 1545 ft lbf/lb-mol R
= 8·3143 kJ/kg-mol K **SI**
and M = molecular weight - mol

Values for air and some commonly employed gases are presented in Table 5.1:

Whichever gas is used, the appropriate value of R for that gas is simply substituted into Equation 4 and the design process is exactly the same.

Table 5.1 Values of Characteristic Gas Constant

Gas	Equation	Molecular Weight M	Gas Constant R - ft lbf/lb R
Air		28·96	53·3
Nitrogen	N_2	28·01	55·2
Oxygen	O_2	32·00	48·3
Carbon dioxide	CO_2	44·01	35·1
Steam	H_2O	18·01	85·8
Argon	Ar	39·95	38·7

2.3.2.1 The Use of Nitrogen

It will be noticed that there is little more than 3% difference between the values of R for air and nitrogen. This is not surprising since about 78% of air, by volume, is nitrogen, and the two constituent gases have very similar molecular weights. As a consequence little error would result if a system in which nitrogen gas was used for conveying a material, was to be inadvertently designed on the basis of air.

If carbon dioxide or superheated steam was to be used to convey the material, however, there would be a very significant error. Gases other than air and nitrogen are often used for specific pneumatic conveying duties.

3 THE INFLUENCE OF PRESSURE

The influence that air pressure has on volumetric flow rate is shown graphically in Figures 5.3 to 5.5 to highlight the influence of compressibility. These are plots of volumetric flow rate, at the reference atmospheric pressure of 14·7 lbf/in^2 absolute, against actual volumetric flow rate.

To simplify the problem an isothermal situation has been assumed in order to isolate the influence of pressure i.e. $T_l = T_o$. Once again this is a linear relationship. A series of lines representing the relationship for different air pressures is given on each graph, and each one illustrates the relationship for a different type of system.

3.1 System Influences

In Figure 5.3 the pressures considered range from 0 (atmospheric) to 12 lbf/in^2 gauge and so is appropriate to low pressure, typically dilute phase, conveying systems. If an air flow rate of 1500 ft^3/min at free air conditions is considered it can be seen from Figure 5.3 that the actual volumetric flow rate of the air at the material feed point, at the start of the conveying line, will be reduced to about 825 ft^3/min if the air pressure is 12 lbf/in^2 gauge.

Pipeline bore is not included at this stage since it is simply the effect of changes in air pressure that are being illustrated.

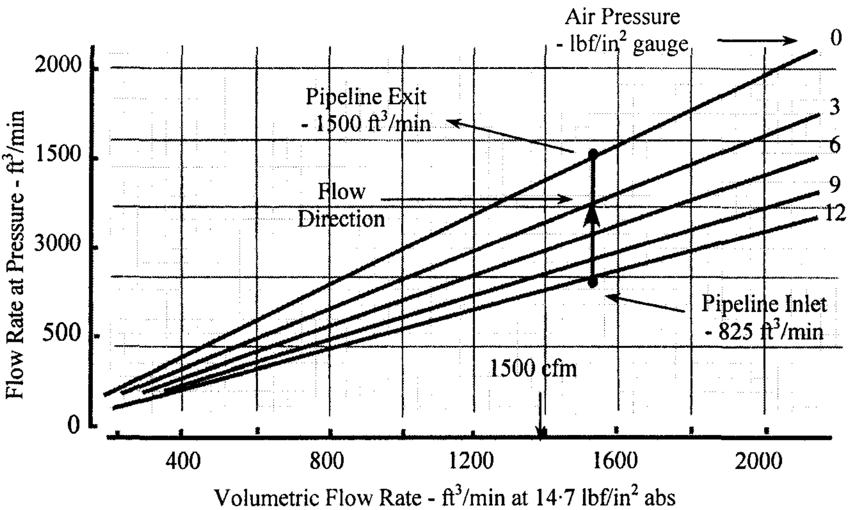

Figure 5.3 The influence of air pressure on volumetric flow rate for low pressure systems.

Alternatively, the flow rate can be determined from Equation 5.8:

$$\dot{V}_1 = \frac{0\cdot0283 \times 519 \times 1500}{(14\cdot7 + 12)}$$

$$= 825 \text{ ft}^3/\text{min}$$

In Figure 5.4 the pressure ranges from 0 to 50 lbf/in^2 gauge and so is relevant to high pressure conveying systems. If the air at the material feed point is at 50 lbf/in^2 gauge, a free air flow rate of 1500 ft^3/min will be reduced to about 340 ft^3/min, as can be seen from Figure 5.4. In both of these cases the air will expand through the conveying line back, approximately, to the free air value of 1500 ft^3/min, at the discharge hopper and filtration unit at the end of the pipeline.

In the case of a vacuum system, free air conditions prevail at the material feed point. The air then expands beyond this and so, if the exhaust is at -8 lbf/in^2 gauge, 1500 ft^3/min of free air will increase to about 3290 ft^3/min, as can be seen from Figure 5.5. Alternatively, the air flow rate can be determined from Equation 5.8 once again:

$$\dot{V}_1 = \frac{0\cdot0283 \times 519 \times 1500}{(14\cdot7 - 8)}$$

$$= 3288 \text{ ft}^3/\text{min}$$

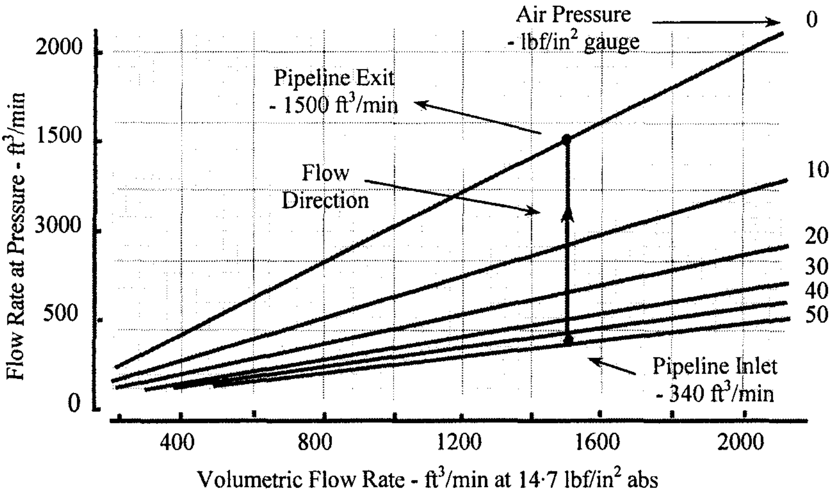

Figure 5.4 The influence of air pressure on volumetric flow rate for high pressure systems.

It can be seen from this range of values that it is extremely important to take this compressibility effect into account in the sizing of pipelines, and particularly so in the case of combined positive and negative pressure systems.

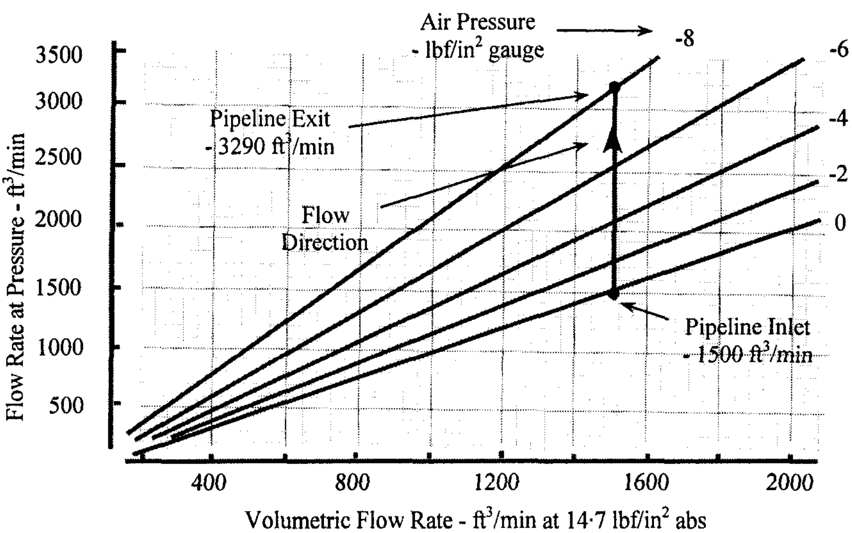

Figure 5.5 The influence of air pressure on volumetric flow rate for negative pressure systems.

An additional point to note is one of the many fundamental differences between positive pressure and vacuum conveying systems. With positive pressure conveying systems the filtration plant can be sized on the basis of the free air flow rate value. For negative pressure conveying systems, however, this is not the case, as will be seen from Figure 5.5.

If the system exhausts at a vacuum of 8 lbf/in^2, for example, the flow rate to be handled by the filter will be about 3290 ft^3/min which is more than double the free air flow rate value, and the filter will have to be sized on 3290 and not 1500 ft^3/min.

3.2 Velocity Determination

If Figures 5.3 to 5.5 are used in conjunction with Figure 5.2, it will be possible to determine the resulting conveying air velocities for given conditions. An alternative to this procedure is to combine the models for actual volumetric flow rate and conveying air velocity.

3.2.1 Working Relationships

From Equation 2 the actual volumetric flow rate:

$$\dot{V}_1 = \frac{\pi d^2 C}{576}$$

and from Equation 7 free air delivered:

$$\dot{V}_o = 35{\cdot}3 \times \frac{p_1 \dot{V}_1}{T_1}$$

and substituting Equation 2 into Equation 7 gives:

$$\dot{V}_o = 0{\cdot}1925 \times \frac{p_1 d^2 C}{T_1} \quad \text{ft}^3/\text{min} \quad - - - - \quad (10)$$

$$= 2{\cdot}23 \times \frac{p_1 d^2 C}{T_1} \quad \text{m}^3/\text{s} \quad - - - - \quad (10\ \text{SI})$$

which is the form required for system design, and rearranging to the form required for checking existing systems gives:

$$C = 5{\cdot}19 \times \frac{T_1 \dot{V}_o}{d^2 \, p_1} \quad \text{ft/min} \qquad - - - - - \qquad (11)$$

$$= 0{\cdot}448 \times \frac{T_1 \dot{V}_o}{d^2 \, p_1} \quad \text{m/s} \qquad - - - - - \qquad (11\ \text{SI})$$

3.2.2 Graphical Representation

It will be seen from these models that a total of five variables are involved and so it is not possible to represent them diagrammatically on a single graph. By neglecting the influence of temperature at this stage the models can be reduced to four variables, and so if particular values of volumetric flow rate are chosen, the influence of the remaining three variables can be shown. This is presented for four values of volumetric flow rate in Figures 5.6 to 5.9, the volumetric flow rates being referred to ambient conditions of temperature and pressure.

These are all graphs of conveying air velocity drawn against air pressure, with pipe bore plotted as the family of curves. The reason for this is that both conveying air velocity and air pressure are infinitely variable in the system, but pipelines are only available in a number of standard sizes. They are drawn once again to illustrate the performance of different types of system. Figures 5.6 and 5.7 cover the range of both positive and negative pressure systems and Figures 5.8 and 5.9 are drawn for positive pressure systems only.

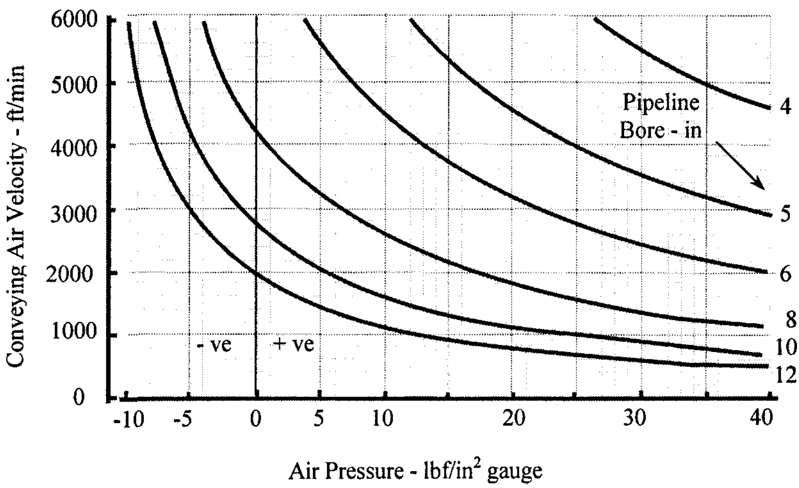

Figure 5.6 The influence of air pressure and pipeline bore on conveying air velocity for a free air flow rate of 1500 ft³/min.

Figure 5.6 clearly illustrates the influence of pressure on conveying air velocity in a single bore pipeline. The slope of the constant pipe bore curves increase at an increasing rate with decrease in pressure. The reason for this can be seen from Equation 11. Conveying line inlet air pressure, p_l, is on the bottom of the equation, and so as its value gets lower, small changes in its value have a more significant effect. This is particularly so for negative pressure systems, and is quite dramatic at high vacuum, as shown on Figure 5.6.

3.2.3 Suck-Blow Systems

On Figure 5.7 the expansion lines for a typical combined positive and negative pressure system are superimposed. This illustrates the problems of both pipeline sizing, with this type of system, and the relative expansion effects at different air pressures.

With 1000 ft^3/min of free air, a 6 in bore pipeline would be required for the vacuum line. This would give an air velocity of about 3560 ft/min at the material feed point and would expand to approximately 4900 ft/min if the exhaust was at -4 lbf/in^2 gauge. If the pressure on the delivery side of the blower was 6 lbf/in^2 gauge, a 5 in bore pipeline would be required. This would give pick-up and exit air velocities of about 3640 and 5130 ft/min respectively.

It will be noted that the pick-up and exit air velocities are very similar for the two parts of the system, but different size pipelines are required.

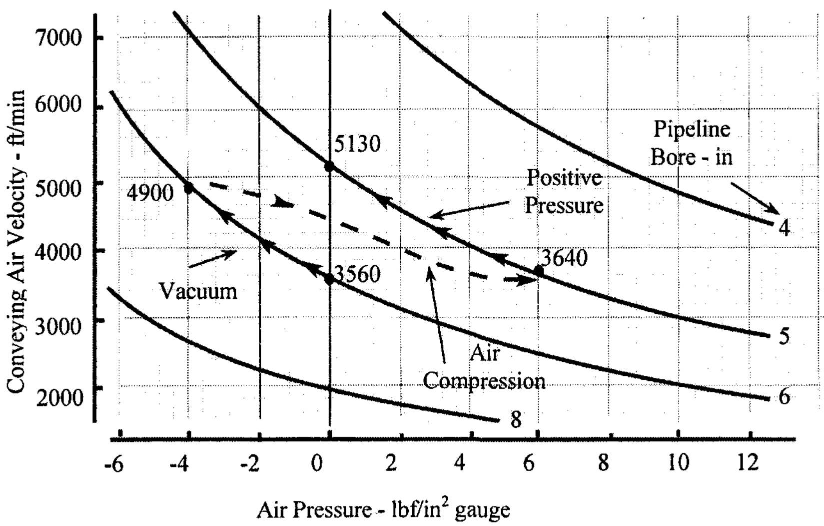

Figure 5.7 Velocity profile for a typical combined positive and negative pressure (suck-blow) system with a free air flow rate of 1000 ft^3/min.

The free air flow rate is clearly the same for the two parts of the system and so it will be seen that it is entirely due to the influence of the conveying line inlet air pressure on the compressibility of the air.

In the above case it has been assumed that the material is conveyed in dilute phase suspension flow and that the minimum conveying air velocity for the material is about 3000 ft/min. If a 20% margin is allowed when specifying a conveying line inlet air velocity, this would need to be about 3600 ft/min.

3.2.4 Low Pressure Systems

In Figure 5.8 a typical velocity profile for a low pressure dilute phase conveying system is shown. In this case the minimum conveying air velocity for the material is approximately 2700 ft/min and so with a 20% margin the conveying line inlet air velocity needs to be about 3240 ft/min. With a free air flow rate of 900 ft^3/min, and a conveying line inlet air pressure of 14 lbf/in^2 gauge, a 5 in bore pipeline would be required.

The resulting conveying line inlet air velocity is about 3380 ft/min, and it will be seen that the air velocity gradually increases along the length of the pipeline as the air pressure decreases. At the end of the pipeline, at atmospheric pressure, the conveying line exit air velocity will be about 6600 ft/min in this 5 in bore pipeline.

The above is simply an example to illustrate the variation in conveying air velocity from feed point to material discharge in a pipeline. For the design of a conveying system Equation 5.10 would be used to evaluate the free air requirements.

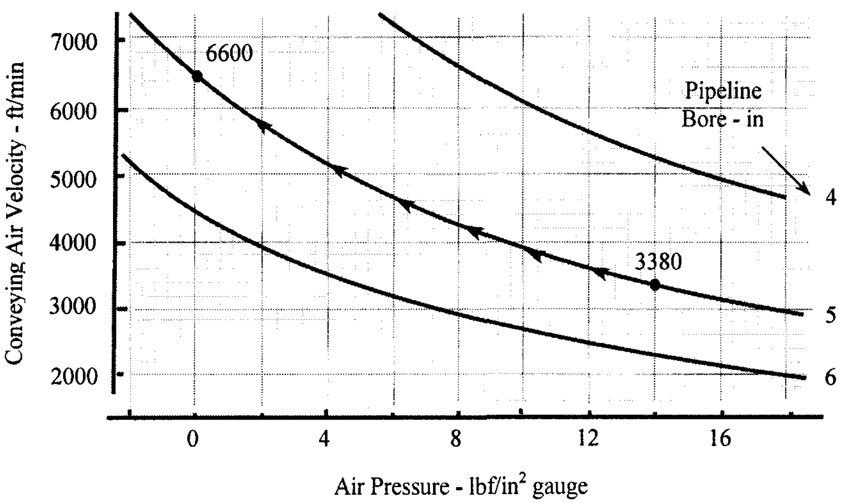

Figure 5.8 Typical velocity profile for a low pressure dilute phase system for a free air flow rate of 900 ft^3/min.

For a 5 in bore pipeline, and with a conveying line inlet air pressure of 14 lbf/in^2 gauge (28·7 lbf/in^2 absolute) and a conveying line inlet air velocity of 3240 ft/min, with air at 59°F, this would come to 862 ft^3/min. If the influence of pressure was not taken into account, and the volumetric flow rate was evaluated on the basis of an air velocity of 3240 ft/min, effectively at the end of the pipeline, the conveying line inlet air velocity that would result at a pressure of 14 lbf/in^2 gauge would be about 1660 ft/min, and the pipeline would almost certainly block.

The influence of air pressure on conveying air velocity is illustrated further with Figure 5.9. This is a plot of conveying air velocity plotted against air pressure, and is drawn for a free air flow rate of 1000 ft^3/min in a 6 in bore pipeline. During the operation of a pneumatic conveying system the conveying line inlet air pressure may vary slightly, particularly if there are variations in the feed rate of the material into the pipeline. If the feed rate increases for a short period by 10%, the conveying line inlet air pressure will also have to increase by about 10% in order to meet the increase in demand.

If the minimum conveying air velocity for the material was 3000 ft/min, and it was being conveyed with a conveying line inlet air pressure of 8 lbf/in^2 gauge, an increase in pressure to only 12 lbf/in^2 gauge would probably be sufficient to result in a pipeline blockage. At low values of air pressure, conveying air velocity is very sensitive to changes in pressure, and so due consideration must be given to this when deciding upon a safety margin for conveying line inlet air velocity, and hence the volumetric flow rate of free air, to be specified for the system.

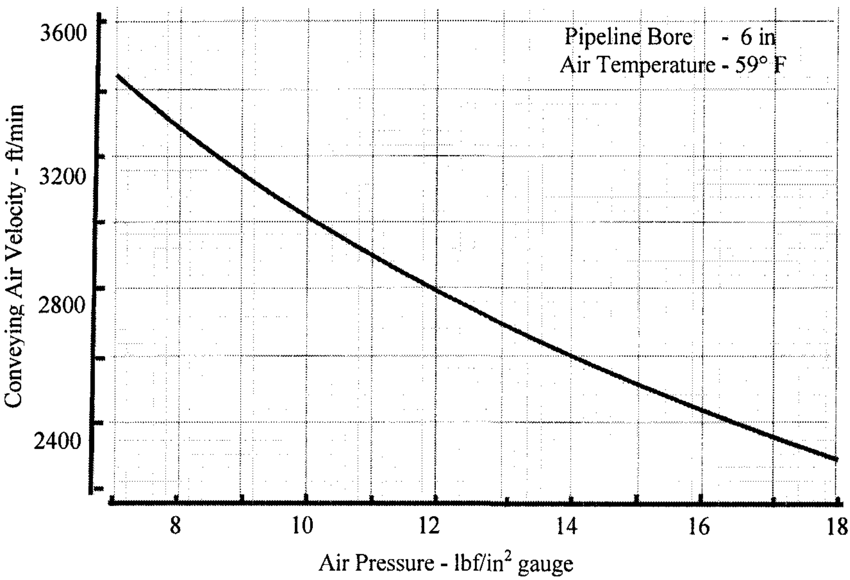

Figure 5.9 The influence of air pressure on conveying air velocity for a free air flow rate of 1000 ft^3/min.

3.2.5 Stepped Pipelines

In the low pressure case illustrated in Figure 5.8 the minimum conveying air velocity for the material was about 2700 ft/min and with a 20% margin this was 3240 ft/min. With the blower available delivering 900 ft^3/min of free air at 14 lbf/in^2 gauge, the resulting conveying line inlet air velocity in a 5 in bore pipeline came to 3380 ft/min. As a pick-up velocity this is quite acceptable but the velocity at the end of the pipeline is quite unnecessarily high at 6600 ft/min.

The velocity profile for a 6 in bore pipeline is also included on Figure 5.8. It can be seen from this that if the pipeline was expanded from 5 to 6 inches at a point in the flow where the pressure was about 5 lbf/in^2 the maximum value of conveying air velocity in the pipeline could be limited to about 5000 ft/min. The Figure 5.8 pipeline is re-drawn in Figure 5.10 with such a step.

From Figure 5.10 it will be seen that the velocity profile has been maintained between very much narrow limits as a result of the step to 6 inch bore. The velocity profile for an 8 inch bore pipeline has also been added and it will be seen that expansion into such a bore would not have been possible. At the step into the 6 inch bore line the velocity drops from 4920 to 3420 ft/min and this is quite acceptable.

Problems arise when the step in bore is incorrectly positioned and the velocity in the larger bore section of pipeline falls below the minimum value for the material. The fact that the velocity at exit from the pipeline is lower than that at entry to the step is of no consequence.

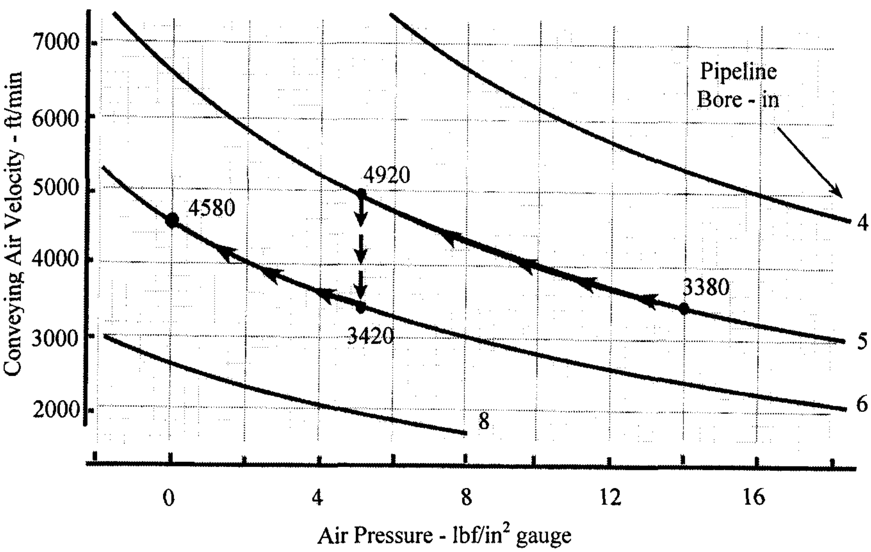

Figure 5.10 The 5 inch bore pipeline velocity profile shown in Figure 5.8 modified by the addition of a step to 6 inch bore.

Stepped pipelines are considered in more detail in Chapter 9. Equations for the evaluation of pressure and velocity are developed and steps for both positive pressure and vacuum conveying systems are considered.

4 THE INFLUENCE OF TEMPERATURE

In the above figures the influence of temperature was not included, so that the influence of pressure alone could be illustrated, and so it was assumed that all flows and expansions were isothermal and at the standard reference temperature. In Equations 7 and 8 the influence of pressure and temperature on actual volumetric flow rate is presented. If the influence of pressure is neglected, in order to separate the effect of temperature, the equation reduces to:

$$\dot{V}_1 = \frac{T_1 \dot{V}_o}{519} \quad \text{ft}^3/\text{min} \qquad (12)$$

The influence that air temperature can have on volumetric flow rate is shown graphically in Figure 5.11. This is a plot of volumetric flow rate at the reference temperature of 59°F, against actual volumetric flow rate at a given temperature. It should be noted that in Equation 12 and Figure 5.11 all pressures are standard atmospheric so that the influence of temperature can be considered in isolation from that of pressure.

It can be seen from Figure 5.11 that changes in temperature do not have the significant effect on volumetric flow rate that changes in pressure can have. This is because the influence of temperature is in terms of the ratio of absolute temperatures and the 460 that has to be added to the Fahrenheit temperature has a considerable dampening effect. Figure 5.11 illustrates the influence of temperature over the range of temperatures from -40°F to 200°F.

Air temperatures higher than 200°F can be experienced, however. Air at a temperature of 200°F will result from the compression of air in a positive displacement blower operating at about 14 lbf/in^2 gauge, and from a screw compressor delivering air at 45 lbf/in^2 gauge it could be more than 400°F. In some cases the material to be conveyed may be at a high temperature and this could have a major influence on the conveying air velocity.

It will be seen from Equation 11 that if the temperature is reduced, then the velocity will fall. This is because the density of the air increases with decrease in temperature. The volumetric flow rate of air that is specified must be sufficient to maintain the desired conveying line inlet air velocity at the lowest temperature anticipated. Due account, therefore, must be taken of cold start-up and winter operating conditions, particularly with vacuum conveying systems which draw in atmospheric air. This point is illustrated quite forcefully in Figure 5.12.

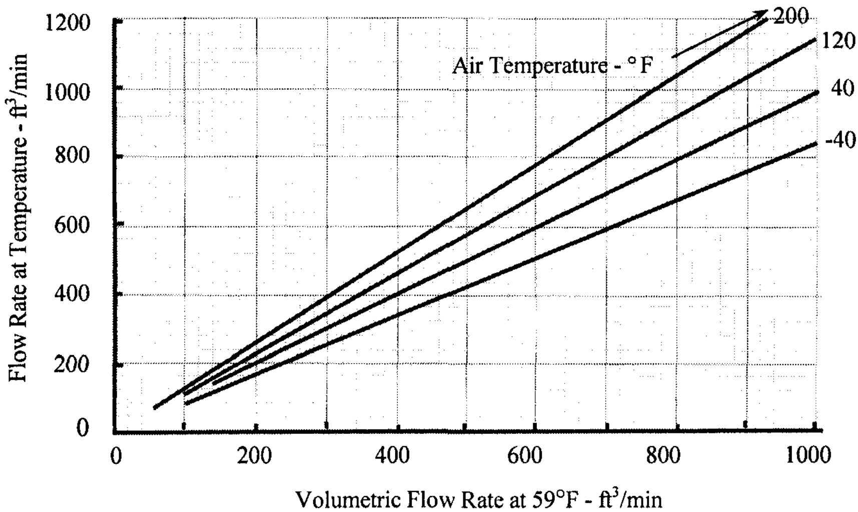

Figure 5.11 The influence of air temperature on volumetric flow rate.

Figure 5.12 is drawn for a 6 in bore pipeline and an inlet air pressure of 15 lbf/in^2 gauge. It will be seen from this that conveying air velocity can be very sensitive to temperature. The average gradient on this plot is about 5 ft/min per °F temperature change, and so if the temperature of the conveying air was reduced for some reason it could result in pipeline blockage in a system operating with a pick-up velocity close to the minimum conveying air velocity for the given material.

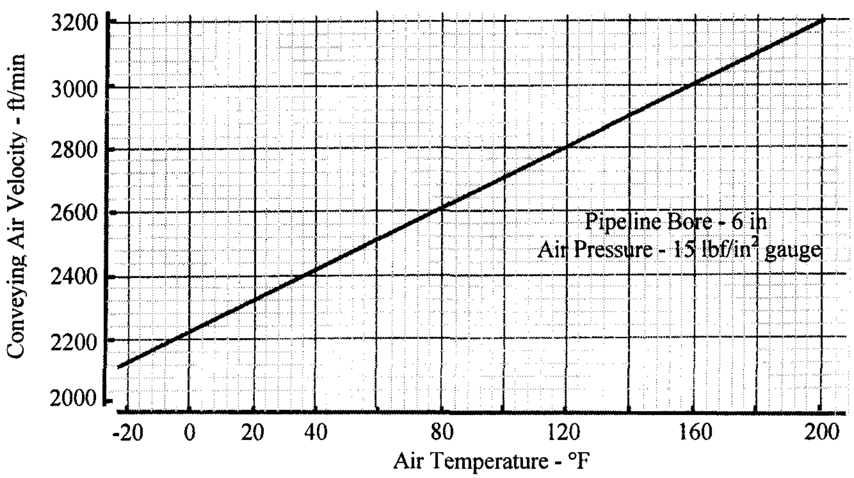

Figure 5.12 The influence of air temperature on conveying air velocity for a free air flow rate of 1000 ft^3/min.

4.1 Conveyed Material Influences

The above analysis refers to the situation with regard to the air only. For the conveying line, however, the material also has to be taken into account, and although the air may be at 60°F, the material to be conveyed may be at 400°F or more. In order to determine the temperature of the conveyed suspension it is necessary to carry out an energy balance. If a control surface is taken around the material feeding device and the immediate pipelines, an energy balance gives:

$$\left(\dot{m}\ Cp\ t\right)_p + \left(\dot{m}\ Cp\ t\right)_a = \left(\dot{m}\ Cp\ t\right)_s \qquad \text{- - - - - -} \qquad (13)$$

where $\dot{m}$ = mass flow rate - lb/h
Cp = specific heat - Btu/lb R
and t = temperature - °F

and the subscripts refer to:
p = conveyed material or product
a = air
and s = suspension

if heat exchanges with the surroundings, kinetic energies and other minor energy quantities are neglected.

It is the temperature of the suspension, t_s, that is required and so a rearrangement gives:

$$t_s = \frac{\dot{m}_p\ Cp_p}{\dot{m}_s\ Cp_s}\ t_p + \frac{\dot{m}_a\ Cp_a}{\dot{m}_s\ Cp_s}\ t_a \quad \text{°F}$$

From continuity

$$\dot{m}_s = \dot{m}_a + \dot{m}_p \quad \text{lb/h} \qquad \text{- - - - - - - -} \qquad (14)$$

and by definition

$$\dot{m}_p = \phi\ \dot{m}_a \quad \text{lb/h} \qquad \text{- - - - - - - - -} \qquad (15)$$

where ϕ is the solids loading ratio of the conveyed material

and

$$Cp_s = \frac{\dot{m}_a\ Cp_a + \dot{m}_p\ Cp_p}{\dot{m}_a + \dot{m}_p} \quad \text{Btu/lb}$$

Substituting these gives:

$$t_s = \frac{\phi \; Cp_p \, t_p + Cp_a \, t_a}{\phi \; Cp_p + Cp_a} \quad {}^{\circ}\mathrm{F} \qquad - - - - \qquad (16)$$

With so many variables it is difficult to illustrate the relationship graphically. One case has been selected, however, for a conveyed material at a temperature of 60°F, having a specific heat of 0·24 Btu/lb R and this is presented in Figure 5.13. This illustrates the influence that conveying line inlet air temperature and solids loading ratio can have on the resulting suspension temperature.

Figure 5.13 relates to the dilute phase conveying of a material with a positive displacement blower, where the conveying line inlet air temperature might be up to about 220°F. This shows that the solids loading ratio has a dominating effect on the suspension temperature, even with dilute phase conveying. Unless the conveyed material has a very low specific heat value, and is conveyed in very dilute phase, the temperature of the conveyed suspension will be close to that of the material to be conveyed. If cold air is used to convey a hot material, therefore, the cooling effect on the material of the cold air will be minimal. This is illustrated in more detail in Figure 5.14 where material and air inlet temperatures of 1000°F and 60°F respectively have been considered.

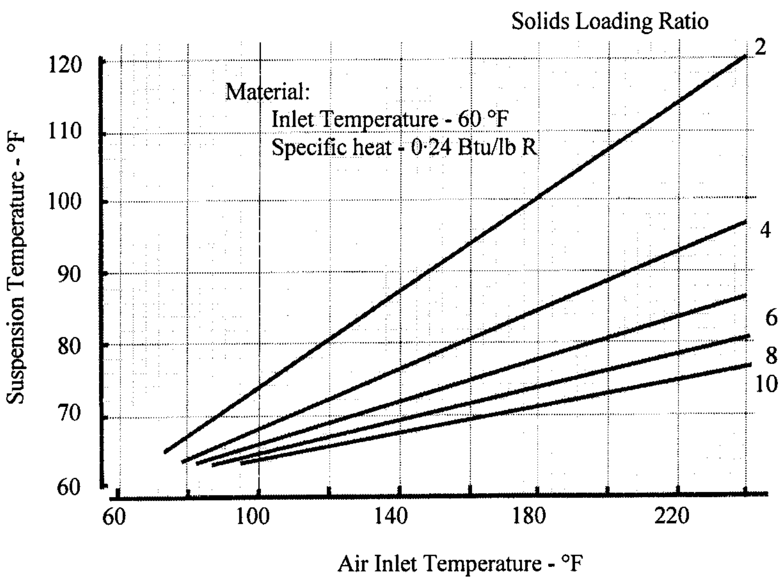

Figure 5.13 The influence of air inlet temperature and solids loading ratio on the equilibrium temperature of the suspension.

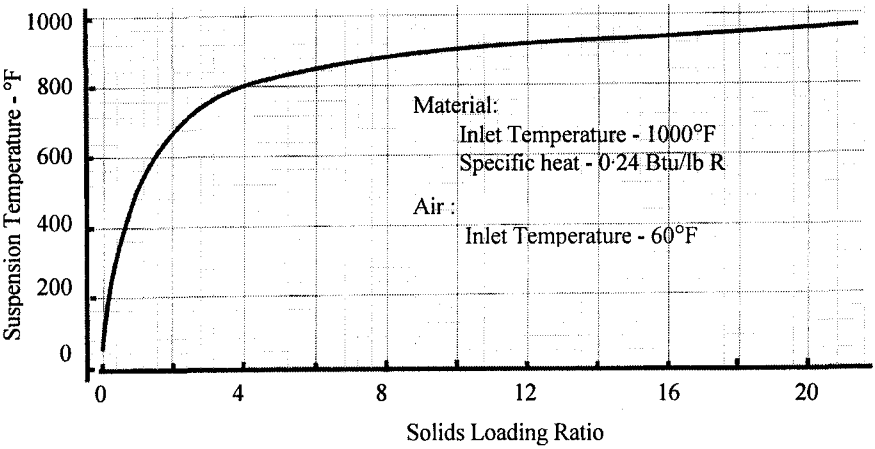

Figure 5.14 Influence of solids loading ratio on the equilibrium temperature of the suspension.

Figure 5.14 is also drawn for a material having a specific heat value of 0·24 Btu/lb R and shows the influence of solids loading ratio. It must be stressed that the suspension of material and air will only reach the equilibrium temperature at some distance from the pipeline feeding point, for thermal transient effects have to be taken into account.

The heat transfer process depends additionally upon the thermal conductivity and shape and size of the particles. It is a time dependent process and with the high velocities required in dilute phase conveying, equilibrium will not be fully established by the end of the pipeline with many materials. Since volumetric flow rate decreases with decrease in temperature, if there is any doubt with regard to the temperature of the air at the start of a conveying line, the lowest likely value should be used for design purposes.

Particular care should be taken with vacuum conveying systems that are required to convey hot materials. There are several points that need to be taken into consideration here. At the material feed point into the pipeline air at atmospheric temperature will generally apply. At any steps in the pipeline, however, the air will be at a significantly higher temperature as a result of the heat transfer. Care must also be exercised with the specification of the exhauster, for this is generally based on the volumetric flow rate of the air drawn into the exhauster.

4.1.1 Specific Heat

Specific heat is clearly an important property in this analysis and typical values are given in Table 5.2 and specific heat values for air and water are also added for reference purposes. That for air is a basic element in the model, of course.

Table 5.2 Typical Specific Heat Values

	Material	Specific Heat Btu/lb R
Metals	Copper	0·09
	Nickel	0·11
	Steel	0·11
	Aluminum	0·21
	Magnesium	0·24
Non-Metals	Sand, dry	0·19
	Firebrick	0·23
	Coal	0·31
	Cotton	0·31
	Bakelite	0·38
	Cork	0·45
Note	Air	0·24
	Water	1·00

It will be noted that water has a very much higher specific heat value than any of the other materials listed, and so if a material has a high moisture content this could have a considerable influence on the specific heat of the material.

5 THE INFLUENCE OF ALTITUDE

As elevation increases, pressure naturally decreases, and so the elevation of a plant above sea level should always be noted for reference. With increase in elevation there is a corresponding drop in the value of the local atmospheric pressure and this will influence many of the velocities and volumetric flow rates in the calculations. There is, of course, a direct influence on the performance of vacuum conveying systems, since any reduction in atmospheric pressure automatically reduces the available pressure difference. The variation of the local value of atmospheric pressure with the elevation of a plant above sea level is presented in Figure 5.15.

5.1 Atmospheric Pressure

Figure 5.15 shows that for a plant located 3000 ft above sea level there is a reduction of more than 10% in atmospheric pressure, and equates to a reduction in pressure of about 1·6 lbf/in^2 or 3·3 in Hg. It will be seen from this that the influence of altitude should be considered in detail for plants located above about 1000 ft, particularly if a vacuum conveying system is to be considered. The normal atmospheric pressure at sea level can fluctuate quite naturally by ± 1 in Hg on a day to day basis, which equates to a change in elevation of about 1000 ft.

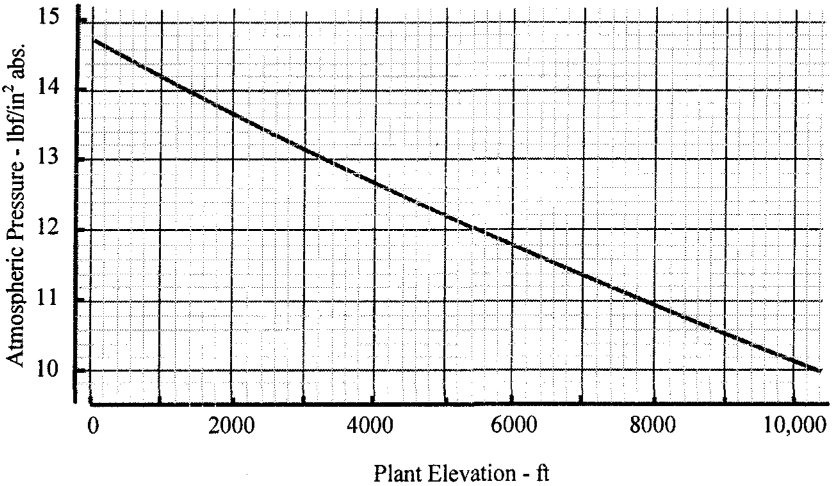

Figure 5.15 The influence of plant elevation on the local value of atmospheric pressure.

6 MOISTURE AND CONDENSATION

Air naturally contains a certain amount of water vapor. The amount of water vapor that air can contain depends upon both temperature and pressure. A decrease in temperature or an increase in pressure can result in condensation occurring when the air passes through the saturation point. The problem with condensation, however, is that it can sometimes be very difficult to predict. The presence of moisture may even be unknown if it cannot be seen, although its effects will certainly be evident.

The addition of water to a bulk solid can have a significant effect on its flowability. Condensation usually occurs on the walls of containing vessels and surfaces such as hoppers, silos and pipelines. Although the effect might be localized, the material/surface interface is critical to the smooth operation of most bulk solids handling plants. Some materials are hygroscopic and will naturally absorb moisture from the air without condensation occurring. For these materials it is generally necessary to dry the air, that comes into contact with the material, to a value of relative humidity below that at which the material is capable of absorbing atmospheric moisture.

Equations are derived and presented that will enable the amount of moisture associated with air to be evaluated. Graphs and charts are also included, to illustrate the influence of the main variables, and to give some idea of the order of magnitude of the potential problem. From the data presented it will be possible to determine rates of condensation and evaporation in processes such as the compression and expansion of air, as well as heating and cooling.

6.1 Humidity

The amount of water vapor that air can support is not constant but varies with both temperature and pressure. Once air is saturated, a change in either temperature or pressure can result in condensation occurring. The terms used here are relative humidity and specific humidity, and the Ideal Gas Law, commonly used for air, provides the basis for modeling moist air.

Specific humidity is the ratio of the mass of water vapor to the mass of dry air in any given volume of the mixture. It is usually expressed in terms of lb of water per lb of dry air. Relative humidity is the ratio of the partial pressure of the vapor actually present to the partial pressure of the vapor when the air is saturated at the same temperature. It is usually expressed as a percentage, with 100% representing saturated air.

Thus, specific humidity is a measure of the moisture content of the air, and relative humidity is a measure of the ease with which the atmosphere will take up moisture. Relative humidity is usually obtained by means of wet- and dry-bulb thermometers, or some other form of hygrometer, and specific humidity can be calculated.

6.1.1 Specific Humidity

Specific humidity, ω, is the ratio of the mass of water vapor to the mass of dry air in any given volume of the mixture:

$$\omega = \frac{m_v}{m_a} \quad lb_v/lb_a \qquad (17)$$

where m_v = mass of vapor - lb
and m_a = mass of air - lb

From the Ideal Gas Law:

$$144\, p_a\, V = m_a\, R_a\, T \qquad (18)$$

At low values of partial pressure, water vapor can also be treated as an Ideal Gas, and so:

$$144\, p_v\, V = m_v\, R_v\, T \qquad (19)$$

where p_a = partial pressure of air - lbf/in^2
p_v = partial pressure of water vapor - lbf/in^2
V = volume of mixture - ft^3
R = characteristic gas constant - ft lbf/lb R
and T = absolute temperature of mixture - R

and note that V and T will be the same for both the air and the vapor, since the two constituents are intimately mixed.

The partial pressure of water vapor, p_v, varies with temperature. For reference, values are given on Figure 5.16. The partial pressure of water vapor increases exponentially with increase in temperature and so the partial pressure axis on Figure 5.16 is split in two. The axis on the right hand side, for high temperature air, is magnified by a factor of ten, compared with that on the left hand side for low temperature air. It will also be seen that at 32°F, the freezing point for water, that a significant quantity of vapor still exists in the air.

At temperatures below 32°F, therefore, water vapor will precipitate as ice onto cold surfaces, without passing through the liquid phase. By the same reasoning, wet surfaces that are frozen can be dried, for the ice evaporates directly into vapor, without the surface becoming wet.

The characteristic gas constants for the two constituents can be obtained from Equation 9 and values for various gases, including steam, are given in Table 5.1. By substituting for R from Equation 9 into Equations 18 and 19 gives:

$$m_a = \frac{144\ p_a\ V\ M_a}{R_o\ T} \quad \text{lb} \quad - - - - - - - - \quad (20)$$

and

$$m_v = \frac{144\ p_v\ V\ M_v}{R_o\ T} \quad \text{lb} \quad - - - - - - - - \quad (21)$$

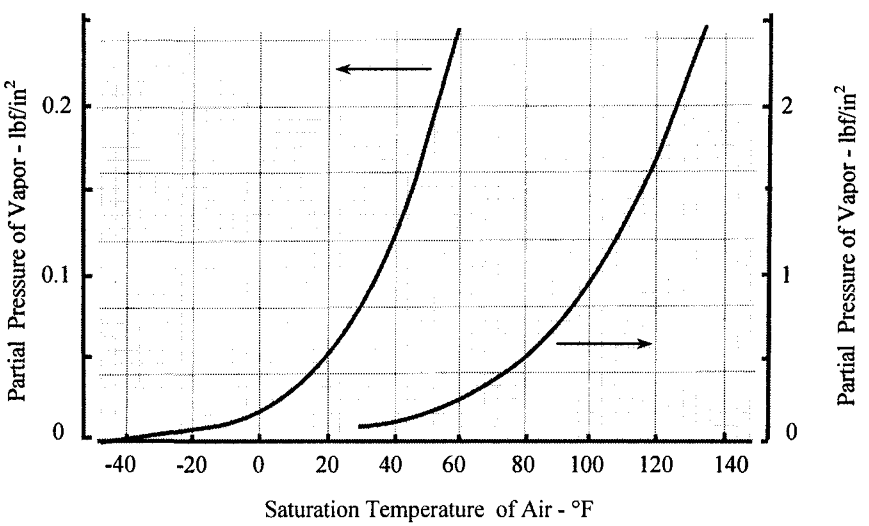

Figure 5.16 The Variation of saturation vapor pressure with temperature.

Substituting Equations 20 and 21 into Equation 17 gives:

$$\omega = \frac{p_v \; M_v}{p_a \; M_a} \quad lb_v/lb_a \quad \text{- - - - - - - - - -} \quad (22)$$

since V and T are common to both constituents.

From Dalton's Law of Partial Pressures:

$$p = p_a + p_v \quad lbf/in^2 \quad \text{- - - - - - - -} \quad (23)$$

where p = total pressure, which for most applications
= atmospheric pressure - lbf/in^2 abs

Thus Specific Humidity, ω, is given by:

$$\omega = \frac{18 \; p_v}{29 \; (p - p_v)} \quad lb_v/lb_a \quad \text{- - - - - - -} \quad (24)$$

Alternatively:

$$\omega = \frac{0 \cdot 622 \; p_v}{p - p_v} \quad lb_v/lb_a \quad \text{- - - - - - - -} \quad (25)$$

6.1.1.1 The Influence of Temperature

A graphical representation of this equation is given in Figure 5.17. This is a graph of the moisture content of saturated air, in pounds of water per 1000 cubic feet of air, plotted against air temperature. This graph is also plotted with a split moisture content axis in a similar manner to Figure 5.16.

The moisture content, in volumetric terms, is obtained simply by multiplying Equation 25 by the density of air. Figure 5.17 is derived for saturated air at atmospheric pressure, which means that this is the maximum value possible for a given value of temperature. It is drawn with two sections, one covering cold air and the other warm air. It will be seen from these that the capability of air for absorbing moisture increases very considerably with increase in temperature.

The moisture content of air can also be expressed in flow rate terms. This is determined simply by using the flow rate form of the Ideal Gas Law, as presented in Equation 5.4, rather than the static form in Equations 18 and 19. Figure 5.18 is such a plot and shows the magnitude of the potential moisture problem, of water associated with air, very well.

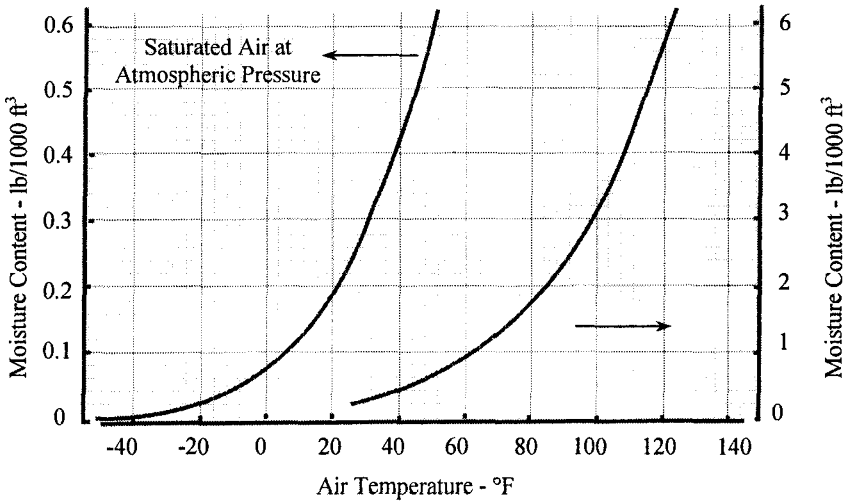

Figure 5.17 The influence of temperature on the moisture content of saturated air.

Figure 5.18 is drawn for saturated air at standard atmospheric pressure and shows how the quantity of water in the air is influenced by both the volumetric flow rate of the air and its temperature.

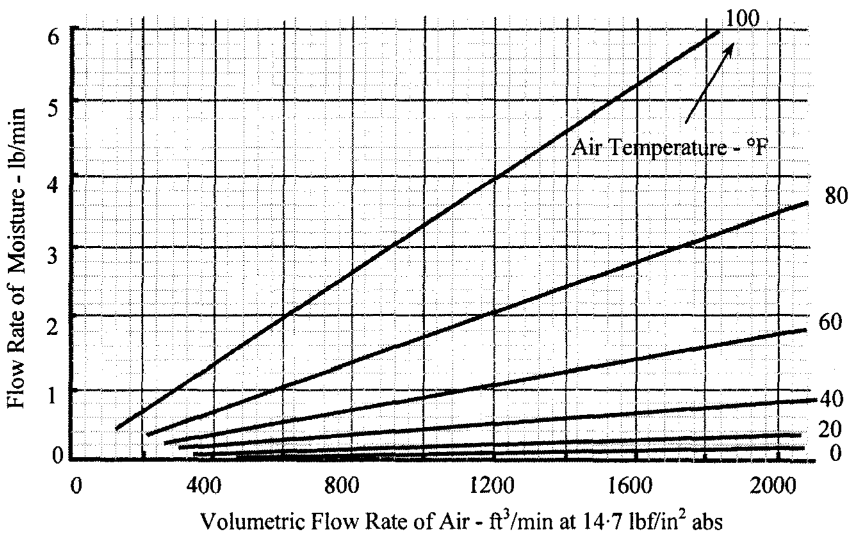

Figure 5.18 The influence of temperature on the flow rate of moisture associated with saturated air.

The influence of the volumetric air flow rate is linear, of course, but that of temperature is not, as illustrated with Figure 5.17. For air at atmospheric pressure this represents the worst case, in terms of the flow rate of water associated with air, since it is drawn for saturated air.

6.1.1.2 The Influence of Pressure

Two further graphical representations of Equation 25 are given in Figures 5.19 and 5.20. These are graphs of moisture content of air, in pounds of water per pound of air, drawn to illustrate the influence of air pressure. Figure 5.19 is a graph of specific humidity plotted against temperature, with lines of constant pressure drawn. The pressures cover a range from -10 to 50 lbf/in^2 gauge and so are appropriate to both positive and negative pressure conveying systems.

Figure 5.20 is a similar plot, but with the x-axis and the family of curves interchanged. Both plots are for saturated air. These show that pressure also has a significant effect on the amount of water vapor that air can absorb, decreasing with increase in pressure. Figure 5.20 shows the influence of pressure on the moisture content capability of air very well, particularly at low pressures and under vacuum conditions.

These plots can be used to determine whether condensation is likely to occur in processes such as the compression and cooling of air. For air that is not initially saturated, however, account has to be taken of the initial relative humidity of the air.

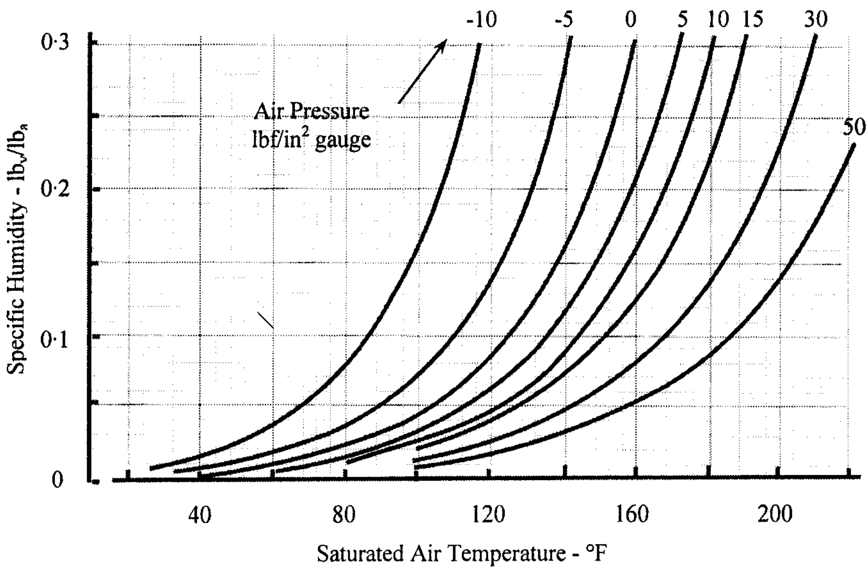

Figure 5.19 The influence of temperature and pressure on the moisture content of saturated air.

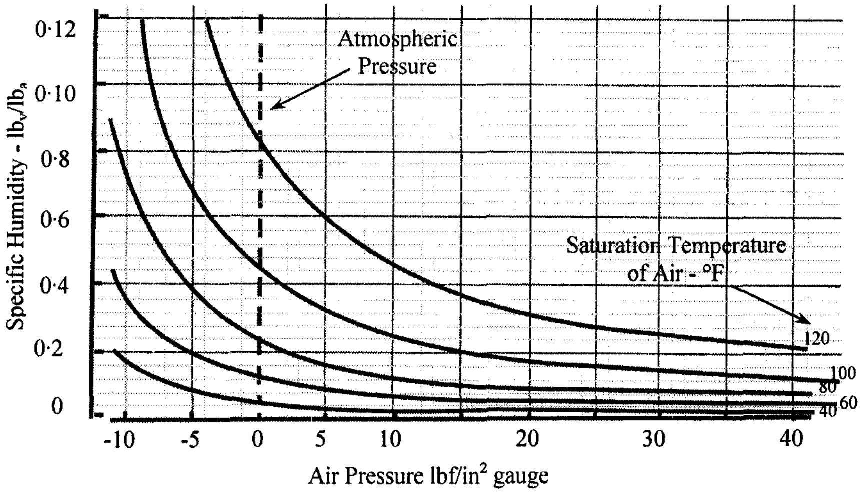

Figure 5.20 The influence of pressure and temperature on the moisture content of saturated air.

6.1.2 Relative Humidity

Relative humidity, φ, is the ratio of the partial pressure of the vapor actually present, to the partial pressure of the vapor when the air is saturated at the same temperature:

$$\varphi = \frac{p_v}{p_g} \qquad (26)$$

where p_v = partial pressure of vapor
and p_g = partial pressure of vapor at saturation

This is usually expressed as a percentage.

This situation can be best represented with lines of constant pressure superimposed on a temperature vs. entropy plot for H_2O. Such a plot is presented in Figure 5.21. This also shows the saturation lines for both liquid and vapor and how these separate the various phases or regions. Air saturated with water vapor, and having a relative humidity of 100%, will lie on the saturated vapor line, g. The vapor in air having a relative humidity less than 100% is effectively superheated steam and so the point will lie in the vapor region.

Point A represents the actual condition of the vapor in the air and it will be seen that it is in the superheated steam region. On the saturation line for the vapor, at the same temperature (point B), the pressure is p_l.

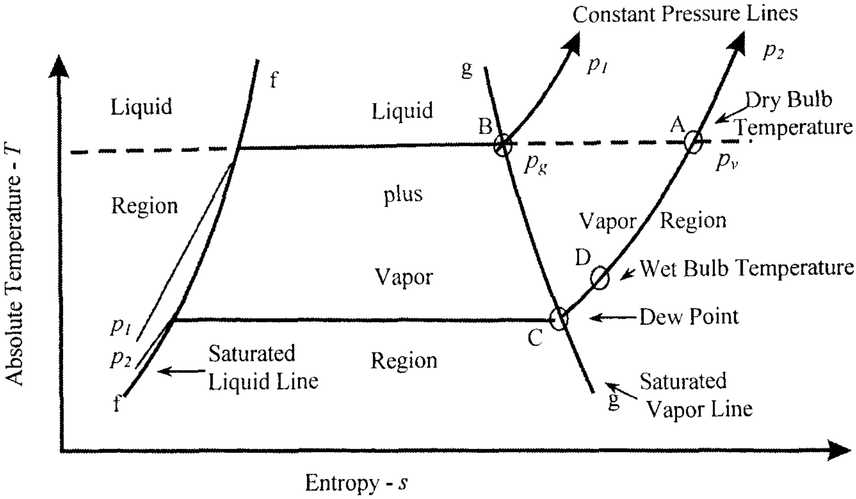

Figure 5.21 Temperature vs. entropy plot for H_2O.

If the air is cooled from point A it will follow the p_2 curve to the saturation line at C, which is the dew point at this pressure. From Figure 5.21 the relative humidity is given as:

$$\varphi = \frac{p_2}{p_1}$$

The pressures p_1 and p_2 can be obtained from Figure 5.16, knowing the corresponding saturation temperatures T_B and T_C.

6.1.3 Psychrometric Chart

The above expression, in terms of pressures, and other equations that can be derived from the Ideal Gas Law, however, are of little practical use in the process of determining relative humidity. For this we generally use wet- and dry-bulb thermometers or a hygrometer. The actual, or dry-bulb temperature, of the air is represented by point B on Figure 5.21 and point D represents the approximate location of the wet-bulb temperature for unsaturated air.

Since this method depends upon equilibrium between heat and mass transfer rates, the equations are rather complicated, and so data is given in charts and tables. The information is usually presented in a psychrometric chart. Such a chart, for air at atmospheric pressure, is shown in Figure 5.22. This is a graph of specific humidity plotted against dry-bulb temperature.

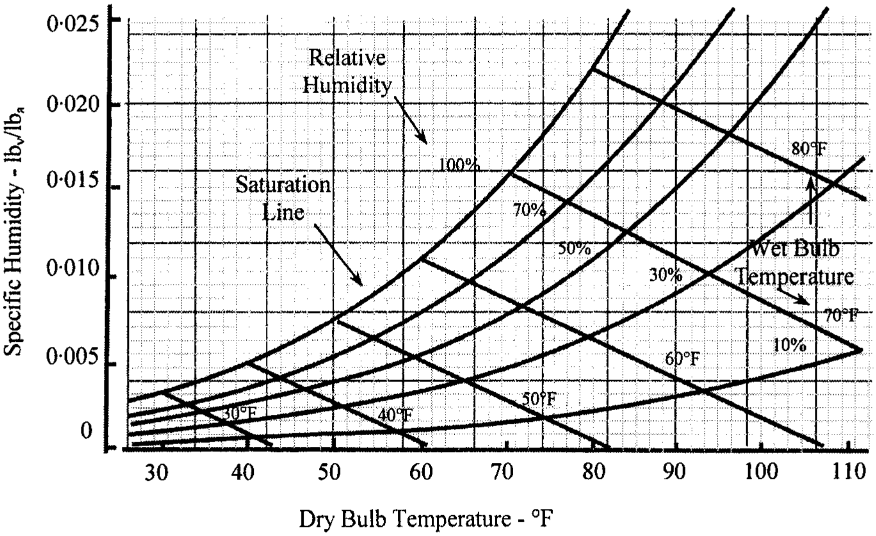

Figure 5.22 Psychrometric chart for air at atmospheric pressure.

The saturation line is presented on this chart and this represents a relative humidity of 100%. This is the same line as that drawn on Figure 5.17. Dry air, or air with a relative humidity below 100%, is represented in the area to the right of the saturation line. Lines of both constant wet-bulb temperature and relative humidity are superimposed on the chart. Thus, if the wet- and dry-bulb temperatures are known, for a given sample of air, both relative humidity and specific humidity can be determined quite simply. On some psychrometric charts lines of constant specific enthalpy and specific volume are also superimposed so that this data can also be obtained quickly if required.

6.1.4 Universal Model

By combining Equations 5.25 and 5.26 an equation is obtained in which both relative humidity and specific humidity appear. This is:

$$\omega = \frac{0 \cdot 622\ \varphi\ p_g}{p - \varphi\ p_g} \quad lb_v/lb_a \qquad (27)$$

Thus, with relative humidity, φ, obtained from a hygrometer, the pressure, p, obtained from a barometer or pressure gauge, and the saturation pressure, p_g, obtained from Figure 5.16 or an appropriate set of tables, the specific humidity of any sample of air can be readily evaluated.

NOMENCLATURE

			SI
A	Pipe section area	in^2	m^2
C	Velocity	ft/min	m/s
Cp	Specific heat	Btu/lb R	kJ/kg K
d	Pipe bore	in	m
m	Mass	lb	kg
$\dot{m}$	Mass flow rate	lb/h	kg/s, tonne/h (1 tonne = 1000 kg)
M	Molecular weight	-	-
p	Pressure	lbf/in^2	kN/m^2, bar (1 bar = 100 kN/m^2)
R	Characteristic gas constant	Btu/lb R	kJ/kg K
R_o	Universal gas constant	Btu/lb-mol R = 1·986 Btu/lb-mol R	kJ/kg-mol K = 8·314 kJ/kg-mol K
s	Specific entropy	Btu/lb R	kJ/kg K
t	Actual temperature	°F	°C
T	Absolute temperature	R = $t + 460$	K = $t + 273$
V	Volume	ft^3	m^3
$\dot{V}$	Volumetric flow rate	ft^3/min	m^3/s

Greek

ρ	Density	lb/ft^3	kg/m^3
ϕ	Solids loading ratio $= \dot{m}_p / \dot{m}_a$	-	-
φ	Relative Humidity	%	%
ω	Specific Humidity	lb_v/lb_a	kg_v/kg_a

Subscripts

a	Air
f	Saturated liquid
fg	Change of phase (evaporation) (= g - f)
g	Saturated vapor
p	Conveyed material or product
s	Suspension
sat	Saturation value or conditions
v	Water vapor
o	Reference conditions (free air)

p_O = 14·7 lbf/in^2 absolute = 101·3 kN/m^2 abs

T_O = 519 R = 288 K

1, 2 Actual conditions - usually inlet and outlet

REFERENCE

1. D. Mills. Optimizing pneumatic conveying. Chemical Engineering. Vol 107. No 13. pp 74-80. Dec 2000.